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TESTING TIDAL TURBINES - PART I: STEADY TOWING TESTS VS. TIDAL MOORING TESTS

P. Jeffcoate, B. Elsaesser & T. Whittaker, *Queen's University Belfast, UK*
C. Boake, *ARR Ltd, UK*

ABSTRACT

Tidal turbines have been tested extensively at many scales in steady state flow. Testing medium- or full-scale devices in turbulent flow has been less thoroughly examined. The differences between turbine performances in these two different states are needed for testing method verification and numerical model validation. The work in this paper documents the performance of a 1/10 scale turbine in steady state pushing tests and tidal moored tests. The overall performance of the device appears to decrease with turbulent flow, though there is increased data scatter and therefore, reduced certainty. At maximum power performance, as velocity increases the difference between the mechanical power and electrical power in steady and unsteady flow increases. The drive train conversion efficiency also decreases. This infers that the performance for this turbine design is affected by the presence of turbulent flow.

NOMENCLATURE

A	Swept area (m^2)
ADP	Acoustic Doppler Profiler
ADV	Acoustic Doppler Velocimeter
C_p	Power coefficient
D	Rotor diameter (m)
HATT	Horizontal axis tidal turbine
P_{flow}	Power in flow (W)
$P_{\text{mechanical}}$	Mechanical power (W)
R	Rotor radius (m)
RPM, n	Revolutions per minute
T_{dynamic}	Dynamic thrust (N)
T_{static}	Static thrust (N)
TSR, λ	Tip speed ratio
U	Flow velocity (m/s)
ρ	Density (kg/m^3)
τ	Torque (Nm)
ω	Rotational velocity (rad/s)

1. INTRODUCTION

With more and more prototype tidal devices being deployed into the water, the reliability of the testing procedures and scale performance results becomes increasingly more important. Scaled devices are typically tested in flume tests, before prototypes are tested in larger scale towing tests; but do these steady flow tests accurately represent the performance of a device in real flows?

Steady flow tests give a good preliminary test condition and an indication of device performance^[1]. In 'real' flows there are the complications of varying inflow velocity, varying inflow direction and turbulence. Current scale tests and numerical models for resource assessment, device performance and array comparisons do not account for this high frequency varying inflow, though they may include typical tidal cycles and increase turbulence intensity to represent turbulent conditions^[2-4]. The accuracy of these tests and models must therefore be examined by testing an identical device in both steady flow conditions and 'real' turbulent tidal conditions.

Queen's University Belfast have the unique ability to be able to test devices in controlled towing test conditions in a lake environment and 'real' tidal flows in a highly turbulent tidal environment, at Strangford Lough Narrows. A novel testing method has been developed by QUB and Wave Barrier Ltd to test medium-scale devices in different conditions to determine the impact of tidal flows on device performance.

This paper details the work conducted under the Invest Northern Ireland funded Tandem Tidal Turbine (INI TTT) project. The novel catamaran testing platform and its use in both lake towing tests and moored tidal tests are presented. The paper then presents the results of a 1/10th scale, 1.5m diameter rotor, horizontal axis turbine when

in steady inflow tests and turbulent tests. The device performance is compared using the power performance and thrust performance curves measured in each condition. The flow velocities used for these criteria were recorded using Nortek Acoustics Doppler Profilers (ADPs) and Acoustic Doppler Velocimeters (ADV)s).

2. EXPERIMENTAL SET-UP

2.1 TIDAL TURBINE

A mono-strut horizontal axis tidal turbine (HATT) was used for testing the effect of turbulence on tidal turbines. The 1.5m diameter (D) turbine is shown in Figure 1:



Figure 1: Mono-strut HATT

The rotor consists of four Eppler E387 wind turbine blades mounted into a hub fairing. These blade profiles are not idealised for tidal testing, but provided a quick and easy off the shelf blade that could be easily replicated in numerical models in future computational testing. The fairing was designed to cover the blade roots, thus reducing flow perturbation and rotation around the complex root geometry.

The rotor drives the central main shaft, which is coupled with the power train. The power train

consists of a torque sensor, a 1:10 gearbox and a generator. The torque sensor, mounted using flexible couplings, measures the rotor RPM and shaft torque. The generator converts the mechanical power to a 3-phase electrical power output^[5]. The current, voltage and electrical power; and the torque, RPM and mechanical power were recorded and used to calculate the mechanical, electrical and hydrodynamic performance of the turbine.

2.2 TESTING CATAMARAN

A catamaran testing platform was developed between QUB and Wave Barrier Ltd to test tidal turbine devices, detailed in Jeffcoate et al^[5]. The catamaran was constructed from four hull sections into a structure of 14.75m by 6.1m, as shown in Figure 2. The test section created was 10.2m by 5.2m. The turbine was suspended from the trusses spanning the test section, so that the shaft depth was 1.5m ($1D$) below the surface, show in Figure 3.

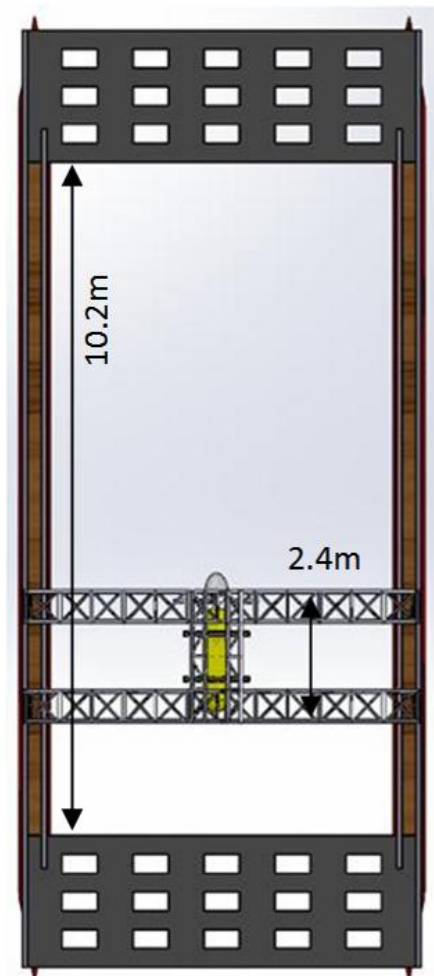


Figure 2: Testing catamaran (Plan view)



Figure 3: Turbine suspended from gantry into water

2.3 PUSHING AND MOORED TESTS

In steady tests, the testing catamaran was pushed in a still-water lake to simulate steady and uniform inflow conditions. This testing method used a second ‘pushing’ catamaran, with a 25HP engine on the stern of each hull coupled to the testing catamaran shown in Figure 4A. The turbine was pushed through the still water and due to the length and width of the rig the directionality of the tests was highly consistent.

During the tidal tests, the testing catamaran was coupled to a mooring catamaran, which had a 4-point seabed mooring attached to trusses spanning the rig. The additional hulls gave extra buoyancy to the rig and increased the distance between the mooring chain and the turbine, thus

reducing the interference between the two. The three coupled catamarans are shown in Figure 4B.

2.4 TESTING ENVIRONMENT

The lake that the steady state pushing tests were conducted in was approximately 6m deep and 33m wide, with a towing track of 400m. The resulting blockage ratio was approximately 1%, though the depth varied throughout the lake area. Most importantly the depth was relatively constant along the towing line resulting in a rotor diameter to depth ratio of 0.25. The full site details can be available in Atcheson^[6]. The inlet velocity, or boat speed, was regulated using the engine throttle. This was maintained at a constant velocity for the length of the towing track, for velocities of 0.8m/s, 1.0m/s, 1.2m/s and 1.4m/s.

The mooring arrangement was located in Strangford Narrows, at approximately 54°22.8784N 005°33.2381W^[7]. The velocities at this location varied from 0m/s to 1.8m/s, with peak velocities only achievable at spring tidal ranges. The tests at flow speeds comparable to those in the lake were used for performance comparisons. The testing period in the lake ran for approximately 5 weeks, from 23rd July to 29th Aug 2013. The tidal tests were conducted in Strangford

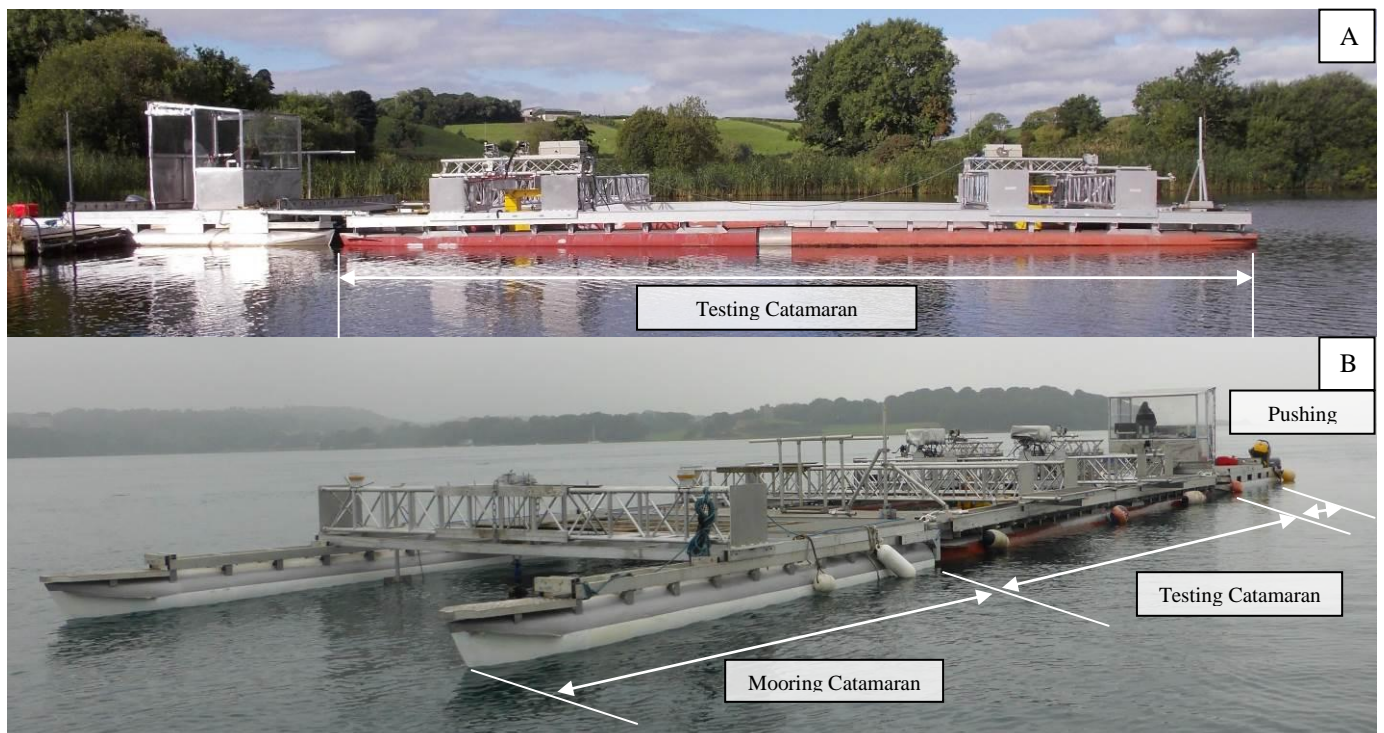


Figure 4: A) Pushing catamaran coupled onto testing catamaran

B) Mooring catamaran with testing and pushing catamarans coupled behind

Narrows after the steady tests, from 26th Sept to 8th Nov 2013. The testing times in the tidal environment were strongly dictated by the tidal range, tide times and daylight hours, so testing was intermittent.

2.5 VELOCITY INSTRUMENTATION

The inflow conditions were measured using a Nortek Aquadopp ADP^[8]. This was mounted down-facing at the bow of the testing catamaran and measured the velocity in 0.2m bands from near the surface to the bed. The inflow conditions were calculated over the rotor area, so the bands that were in the range 0.75m to 2.25m from the surface were depth-averaged. The data was recorded at 0.1s intervals, 10Hz, over an entire run length, which varied from 3 to 4min in the lake and in the tidal site was 3min. This gave between 1200 and 1800 samples per run.

A second ADP was mounted at the bow of the mooring catamaran for the tidal tests. This gave an opportunity to compare the velocities up to 3D laterally and longitudinally spaced.

A Nortek Vector ADV^[9] was also used for point measurements in the tidal flow to calculate the site's turbulence characteristics. Two ADVs were used: the first was mounted in front of the turbine, to capture the inflow conditions, and the second was mounted at various downstream locations. The sample volume was 0.75m below the surface, at the topmost location of the blade tip. The sensor records at a sampling rate of 64Hz for a velocity range of 2m/s. The data from the ADVs can be used to calculate the turbulence intensity and velocity variation with time.

3. PERFORMANCE CALCULATION

The turbine's performance in each environment was derived from the data collected from the ADP and the turbine itself. The inflow velocity was power-weighted averaged across the rotor depth for each run, as per IEC standards^[10]. The

hydrodynamic performance of the turbines was calculated using the power coefficient, C_p .

The power available in the uniform flow, P_{flow} , over the swept area of the rotor, A , was calculated using Equation 1 below, where ρ is the water density (1000kg/m³ fresh water, 1025kg/m³ fresh water) and U is the depth-averaged, time-averaged velocity.

$$P_{flow} = \frac{1}{2} \rho A U^3 \quad (1)$$

The inflow velocity used was the resultant velocity, but the depth-averaged flow angle had an average variation of <1% from the streamwise direction. Since the accuracy of the ADP is 0.6% the difference between the resultant velocity and the streamwise velocity was considered within experimental error and therefore negligible.

The mechanical power, P_{mech} , generated by the turbine was calculated using the torque, τ , measured in Nm and the revolutions per minute (RPM), n :

$$P_{mech} = \tau \frac{2\pi n}{60} \quad (2)$$

The power coefficient is the difference between the power available in the flow and the power extracted by the turbine, hence it is defined as:

$$C_p = \frac{P_{mech}}{P_{flow}} \quad (3)$$

In order to compare the experimental values with inflow velocities, the rotational speeds are normalised into tip speed ratio using the time-averaged inflow velocities. The tip speed ratio, λ , is defined using the rotor radius, R , the rotational velocity, ω , in rad/s, and the inflow velocity:

$$\lambda = \frac{R\omega}{U} \quad (4)$$

The power curves typical of device performance analysis, C_p - λ , could then be derived.

4. RESULTS

4.1 POWER PERFORMANCE

The power coefficient, C_P , to tip speed ratio, λ , curves for the turbine in steady and unsteady flow are shown in Figure 5. The results show the time-averaged value for a range of resistance settings and inflow speeds, with a polynomial best-fit line to show the curve trend.

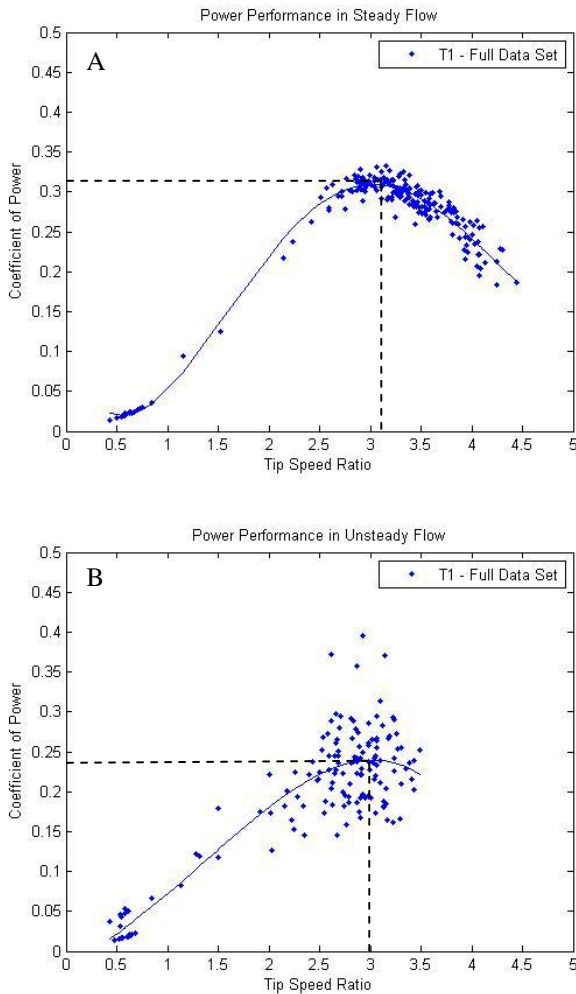


Figure 5: Coefficient of power curves for steady flow (A) and unsteady (B) flow

The steady flow C_P - λ curve shows that the turbine performs with maximum C_P occurring at approximately (3.1, 0.31). Most tidal turbines operate at higher tip speed ratios and higher efficiencies; however, this turbine has wind turbine blades and is therefore not designed for higher density flows. The chord length is relatively short for tidal stream turbines and the blade thickness of 9.1% is also very small. Furthermore, the blades are relatively flexible, thus the lower performance was expected.

However, the data nicely confirms an acceptable performance of the whole system, even without a specific power control. Thus the turbine can be used as a comparative tool between steady and unsteady flow, so the optimum blade design is not required for this experiment.

The power curve is also apparent in the unsteady flow condition, but there are some key differences between the steady and unsteady power curves. Whilst the maximum power coefficient in the steady flow is 0.31 and occurs at a tip speed ratio value of 3.1, the unsteady maximum value is 0.24 at $\lambda=3$. This shows that the maximum power extractable from the flow in a tidal condition can be 23% less than in a steady flow. This value is rather uncertain, however, due to the large amount of scatter in the tidal data.

At optimum C_P the scatter in the steady data is only $\pm 6\%$ so the result is very accurate. In the unsteady, tidal data the scatter can be up to $\pm 40\%$. The C_P - λ curve is highly dependent on the inflow velocity value, shown in Equation 1, so the high amount of scatter may be caused by the large amount of variation in the inflow velocity in the tidal tests.

4.2 VARIATION OVER TIME

The time traces of the torque, RPM and mechanical power for an example steady run and example unsteady run are shown in Figure 6. The average velocity and resistor setting are the same for each run, so direct comparison can be made.

The differences between the steady and turbulent time traces can clearly be seen. The variation in the turbulent time traces is significantly greater than in the steady traces, indicating that the varying inflow conditions have a large effect on the device response and performance.

Full analysis of the velocity variation should be conducted on these different flows, an example of which will be presented in Testing Tidal Turbines Part II – Flow characterisation using ADV data. The differences these flows create in the turbine performance characteristics will be further discussed below.

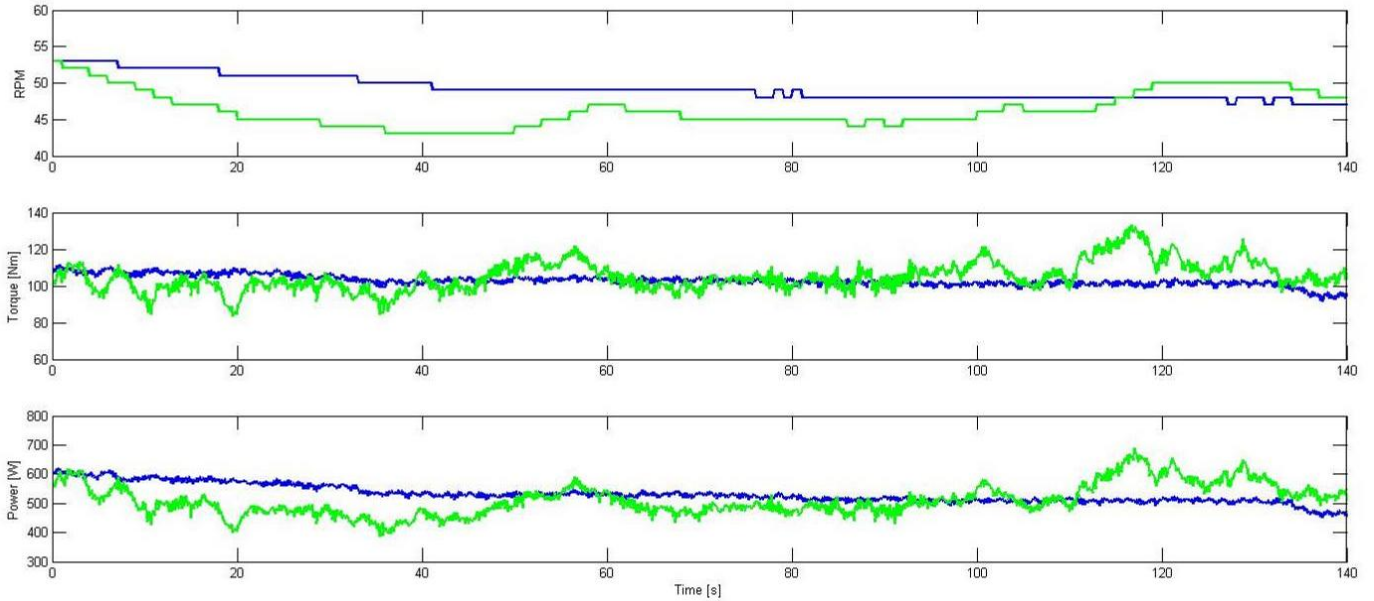


Figure 6: Time traces for steady and unsteady conditions with the same average velocity (blue – steady, green – turbulent)

The resulting time traces for the rotational speed, torque and power output in steady and turbulent flows are rather different. The smaller and more sporadic velocity oscillations that occur in steady flow towing tests lead to fairly constant RPM, torque and power. The variations in the velocity are so small they do not appear to affect the turbine RPM and torque. Longer term variations, for example from 0s to 50s the RPM reduces slightly, are apparent but the shorter period variations are not translated through the system. The resulting standard deviation of the RPM, torque and power are 0.5 (1% of the mean RPM), 1.5Nm (1% of the mean torque) and 11.5W (2% of the mean power) respectively.

In the unsteady flow there is much more variation from the mean. The RPM appears to vary significantly over the time trace and the torque and power fluctuate about the mean at very small time periods. The standard deviations for these traces are 1.7 (3% of the mean), 6.3Nm (6% of the mean) and 46.3W (9% of the mean) for the RPM, torque and power respectively. The highly varying flow, therefore, appears to significantly affect the performance of the turbine. To determine the reason for this analysis of the flow conditions, such as the turbulence intensity, is required. This can be done through assessment of the flow using the ADV data collected. This data is discussed in Part II, but the following analysis in this paper uses time-averaged values, regardless of temporal variations in the turbine performances.

4.3 COMPARATIVE PERFORMANCE

The turbine performances at a full range of resistance values and inflow conditions were shown in Figure 5. These results included all of the data, even those where performance was low. This could be due to, for example, excessively large resistance values. In order to remove most of the scatter and focus on the results at maximum performance the data was cropped so that the top 60% of the coefficient of power only were used. The data that has been included in the subsequent analysis is shown in Figure 7.

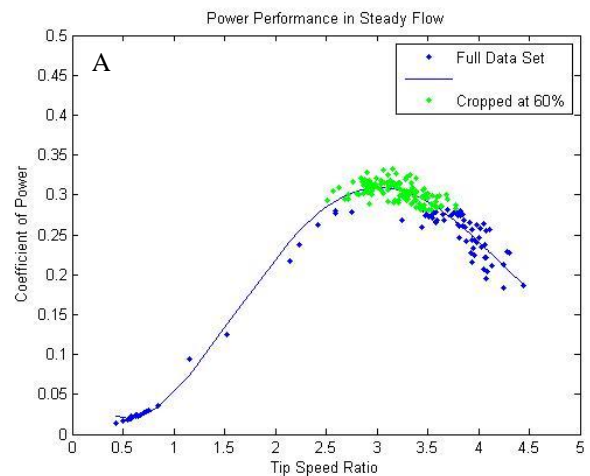


Figure 7A: Steady power curves with data in top 60% highlighted

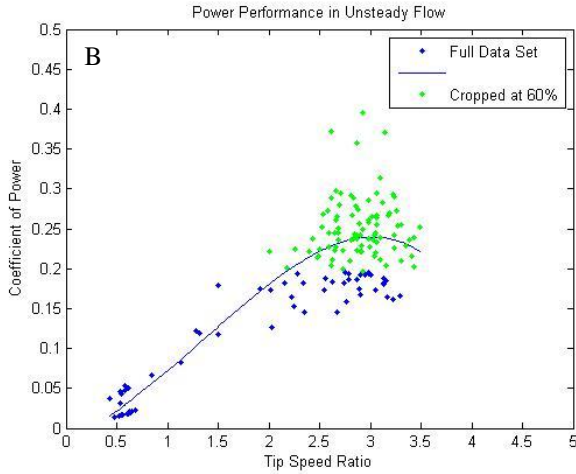


Figure 8B: Unsteady power curves with data in top 60% highlighted

The cropped data was used for comparing the performances at maximum C_P values. Figure 8 shows the comparative C_P - λ curves for steady and unsteady flow. The reduced power extraction in unsteady, tidal flows is clearer here, where the lower C_P values are more evident.

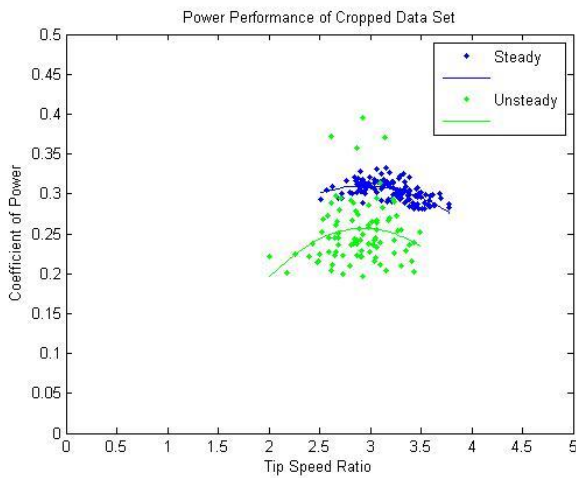


Figure 9: Steady and unsteady power curves for cropped data set

Some of the data points for unsteady flow do produce as much power performance as the turbine in steady flow; however, the higher uncertainty and variation in the results shows that the overall performance may be reduced. The cropped data sets also show that the maximum steady data points occur at higher tip speed ratios than the tidal data set. The TSR varies from 2.5 to 3.7 in steady, whereas in turbulent flow it varies from 2 to 3.5. This indicates that the average rotation speed in steady flows is higher when compared to the average inflow velocity in turbulent flows. This may be due to the higher

variation of the RPM in turbulent flows, as seen in Figure 6. The effect of this flow feature is still to be fully investigated. The velocity, RPM and torque all vary significantly with unsteady inflow conditions, so the resulting means could be quite different. Since C_P is proportional to torque and RPM and inversely to the cube of the velocity, these will all greatly affect the performance curve if there is any variation in the mean values due to temporal variations.

4.4 MECHANICAL PERFORMANCE

The mechanical performance for the data at maximum power performance was assessed. The comparison of the mechanical power against inflow velocity for each of the inflow conditions is shown in Figure 9.

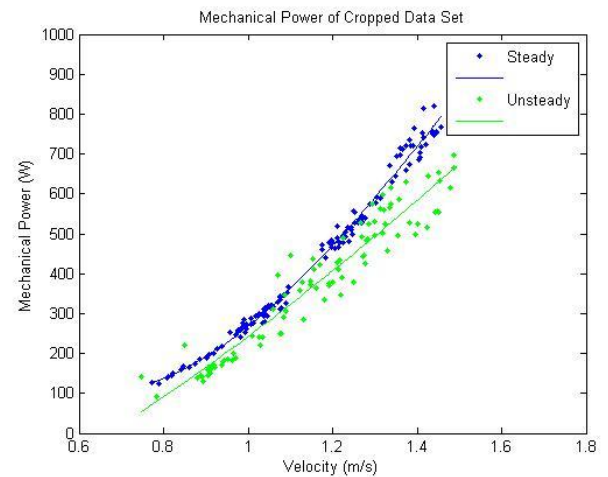


Figure 10: Steady and unsteady mechanical power against inflow velocity

The relationship between the mechanical power and the velocity follows the power law expected from Equations 1 and 3. The power coefficient values are approximate to the maximum value; therefore, the mechanical power is proportional to the power in the flow and thus the cube of the inflow velocity. The relationship is very consistent for the steady flow, due to the small amount of scatter in the C_P values. The unsteady, turbulent results also follow this trend, but the higher amount of C_P data scatter results in some variation from the overall data trend. There is also reduction in the mechanical power output, possibly due to the temporal variations observed earlier. The mechanical performances in the unsteady tests are approximately 40W (14%) less than the steady tests at 1m/s (lower flow speeds) and 130W (18%)

less at 1.4m/s (higher flow speeds). The scatter in the results also increases with velocity, with $\pm 20W$ from the mean at 1m/s and $\pm 60W$ at 1.4m/s.

4.6 ELECTRICAL PERFORMANCE

This trend can also be seen in the electrical performance, in Figure 10, although the difference between the steady and unsteady electrical power outputs has increased. At 1m/s the unsteady flow produces 20% less power than the turbine in steady flow. At 1.4m/s 30% less power is produced. The scatter at 1.4m/s has also increased, with the flow varying up to $\pm 100W$ for the unsteady flow condition. This is shown in Table 1.

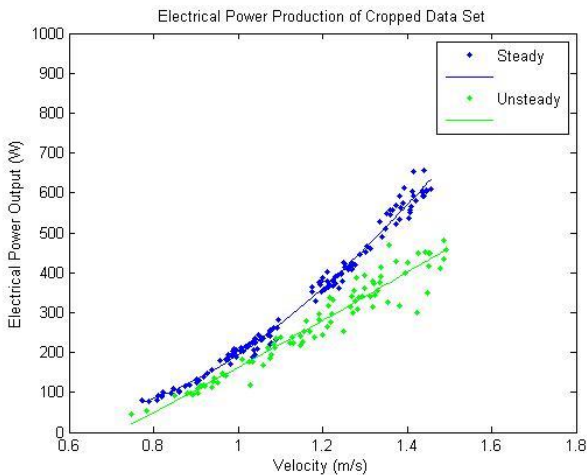


Figure 11: Steady and unsteady electrical power against inflow velocity

Table 1: Difference in power between steady and unsteady flow conditions at two example flow speeds

Power	Flow Speed (m/s)	Power Deficit (%)
Mechanical	1	14
	1.4	18
Electrical	1	20
	1.4	30

As the velocity magnitude increases, the difference between the mechanical power outputs in steady and unsteady flow increases. In addition, as the velocity increases the power conversion efficiency deteriorates. The fluctuating velocity at higher speeds, therefore, leads to reduced power translation through the drive shaft assembly.

5. CONCLUSIONS

The following conclusions have been drawn from the results presented here.

- A comparative testing tool has been designed and it operates across the entire performance range in the tidal flows experienced.
- The turbine can operate up to a maximum C_p of 0.31 when tip speed ratio is approximately 3.1 in steady flows.
- In turbulent flows this is reduced to 2.4 at TSR of 3. This represents an approximate 24% reduction in performance, though there is increased uncertainty due to the levels of scatter.
- There are higher temporal variations in the inflow velocity in the tidal flow, resulting in higher variations in the RPM, torque and power of the turbine.
- Full inflow velocity characterisation needs to be performed to fully understand how the turbine reacts to velocity fluctuations.
- The mechanical and electrical power outputs follow the expected power curve.
- As velocity increases, scatter in the data and differences between the steady and unsteady results increases.
- The electrical power output can reduce by up to 30% (at 1.4m/s) when operating in unsteady, tidal flows compared to steady flows.
- The power conversion efficiency decreases with inflow velocity.
- Full inflow velocity analysis must be conducted using ADV data for accurate comparison of the flow regimes.

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